



Thermodynamic optimization of a regenerator heat exchanger

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ABSTRACT

This paper introduces a general computational model for regenerators fed by a hot fluid stream on one side, whereas the other side is a fluid mixture, in which one of the components (refrigerant) undergoes a phase change. A simplified physical model, which combines fundamental and empirical correlations, and principles of classical thermodynamics, mass and heat transfer is developed and the resulting three-dimensional differential equations are discretized in space using a three-dimensional cell centered finite volume scheme. The proposed model is used to simulate numerically the behavior of the regenerator working under different operating and design conditions. Mesh refinements are conducted to ensure the convergence of the numerical results. The proposed methodology is shown to allow a coarse converged mesh for all simulations performed, therefore combining numerical accuracy with low computational time. The model was then used to determine optimal parameters for a specific geometric configuration of the regenerator, finding that the two-way maximized efficiency operating with optimal porosity and dimensionless mass flow rate of liquid ammonia, is $\eta_{\max,\max} = 45.5\%$ for the optimal pair $(\phi, \dot{m}_s)_{\text{opt}} = (0.5, 0.01)$. As a result, the model is expected to be a useful tool for simulation, control, design, and optimization of regenerators for heat driven refrigerators.

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1. Introduction

Utility companies worldwide acknowledge that heating, ventilation, air conditioning and refrigeration (HVAC-R) systems are responsible for roughly 30% of total energy consumption, including the domestic, commercial and industrial sectors [1], therefore unquestionably with a major impact on energy demand. Researchers in many countries have been involved in developing refrigeration systems that deal with the drawbacks of conventional systems.

Economic and environmental considerations brought a new point of view about refrigeration supplied by renewable heat sources. The international refrigeration industry has been investing considerable resources in that direction [2–13]. For example, an absorption system or ejector cooling system driven by waste heat designed adequately could supply the requirement of oil platforms or refineries in regard to refrigeration and air conditioning.

Absorption and ejector refrigeration systems can compete with compression technology in some applications. Also due to the use of cogeneration systems with turbine and gas engines in industrial

sectors, interest has increased in cooling machines which use residual heat [14]. Development of these machines requires high efficiency in the heat and mass transfer processes which take place in the absorber, which in turn is the most important component in these systems [15,16].

Constructal theory originated within the field of heat transfer [17,18] and has grown into a branch of thermodynamics that deals with engineering design. System configuration is obtained from minimization of resistance (to flow, to heat transfer). In this paper the regenerator is thermodynamically optimized as a result of the balance of competing factors along the lines of other previous studies [19].

A number of papers related to thermodynamic optimization of regenerator heat exchangers can be found in the literature [20–32]. Devices studied include rotary [20–24] or periodic-flow [25,26] regenerators, moving bed heat exchangers [27], and their applications to gas turbines [28] and Stirling engines [29], as well as Stirling-cryogenic [30] and pulse tube refrigerators [31,32]. These optimization efforts involved the determination of the following optimal values and objective functions: optimum channel length and operating period for maximum second-law efficiency [20], and how the latter is affected by heat conduction in the regenerator matrix and unbalanced heat-capacity rates [21]; optimum operational conditions for maximum thermal effectiveness of an air-to-

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Nomenclature

A	area, m^2
A_{gl}^{m}	heat transfer area between matrix and hot fluid at the cell “m”, m^2
c	specific heat, $\text{J kg}^{-1} \text{K}^{-1}$
c_p	specific heat at constant pressure of hot fluid, $\text{J kg}^{-1} \text{K}^{-1}$
c_v	specific heat at constant volume of hot fluid, $\text{J kg}^{-1} \text{K}^{-1}$
c_{pr}	specific heat at constant pressure of internal fluid, $\text{J kg}^{-1} \text{K}^{-1}$
c_1	ratio between the diameter of innermost tube to hot fluid tube
c_2	ratio between the diameter of inner tube to hot fluid tube
D	diameter of external tube (hot fluid), m
d	diameter, m
e_4	thickness of innermost tube, m
h	heat transfer coefficient by convection, $\text{W m}^{-2} \text{K}^{-1}$
h_t	heat transfer coefficient by convection between cold fluid and internal tube, $\text{W m}^{-2} \text{K}^{-1}$
h_{fg}	enthalpy of vaporization of ammonia, J kg^{-1}
k	thermal conductivity of metallic matrix, $\text{W m}^{-1} \text{K}^{-1}$
k_s	thermal conductivity of innermost tube, $\text{W m}^{-1} \text{K}^{-1}$
L	length of heat exchanger, m
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
$\tilde{m}$	dimensionless mass flow rate of hot fluid
$\tilde{m}_s$	dimensionless mass flow rate of liquid ammonia
n_{cell}	number of cells
p_{in}	pressure of liquid ammonia that enters in the regenerator, bar
q	heat transfer rate or enthalpy transferred, W
q_{enth3}	energy rate through the system 3 at cell “m”, W
q_{enth4}	energy rate through the system 4 at cell “m”, W
q_{in}	enthalpy transferred from cell “m” to cell “m – 1”, W
q_{out}	enthalpy transferred from cell “m” to cell “m + 1”, W
$\dot{Q}$	heat transfer rate, W
$\dot{Q}_{\text{m}}$	heat transfer rate received by boiling ammonia, W
r	fraction mass of water in strong solution
T	temperature, K
$\bar{T}$	dimensionless temperature
t	time, s
U_b	ratio between global heat transfer coefficient between system 3 and system 4 while phase changing to global heat transfer coefficient
U_s	global heat transfer coefficient between system 3 and 4 when ammonia is liquid or vapor, $\text{W m}^2 \text{K}^{-1}$
U_{34}	global heat transfer coefficient between system 3 and system 4, $\text{W m}^{-2} \text{K}^{-1}$
u	velocity of refrigerant, m s^{-1}
V	volume, m^3
x_{ref}	length of reference, m
x_r	quality of ammonia
x_r^{m}	quality of ammonia in the cell “m”
$x_r^{n_{\text{cell}}}$	quality of ammonia in the last cell
$\tilde{x}$	dimensionless length

Greek symbols

Δt	amount of time that the fluid stays n the cell “m”, s
Δx	length of cell “m”, m

ε	tolerance
ϕ	porosity
η	efficiency
ρ_g	density of metallic matrix, kg m^{-3}
ρ_r	density of fluid refrigerant, kg m^{-3}
ρ_{NH_3}	density of ammonia, kg m^{-3}
$\rho_{\text{H}_2\text{O}}$	density of water, kg m^{-3}
ρ	density of hot fluid, kg m^{-3}
v	specific volume, $\text{m}^3 \text{kg}^{-1}$
Subscripts	
abs	absorber
ar	wire
cond	condenser
cond,a	conduction between metallic matrix of cell “m” and metallic matrix of cell “m – 1”
cond,p	conduction between metallic matrix of cell “m” and metallic matrix of cell “m + 1”
g	metallic grid
gl	interface between matrix and hot fluid
gs	transversal interface between two cells
H	high
H_2O	water
in	inner tube or inlet
in,4	initial state of liquid ammonia
l	liquid ammonia
L	low
max	maximum
NH_3	ammonia
NH_3,s	liquid ammonia or vapor of ammonia completely vaporized
NH_3,v	vapor of ammonia vaporized accumulate
opt	optimum
pr	internal fluid at constant pressure
r	cold fluid
ref	reference
S	section interface of innermost tube
s	liquid ammonia
s,0	initial state of ammonia in the inner tube
s,in	inlet state of ammonia in the inner tube
sat	saturated
T	refrigerant inside of innermost tube
T	total
t	internal tube
tl	interface between system 1 and system 3
tl4	lateral interface of innermost tube
V	empty space in system 1
v	ammonia vapor
vr	internal fluid at constant volume
w	ammonia in the innermost tube
w,in	inlet ammonia that does not vaporize
0	initial state
0	ambient
3	interface between system 3 (non-vaporized ammonia) and innermost tube
4	interface between innermost tube and liquid ammonia
∞	reference for temperature

Superscripts

m	position of the cell (control volume) in the regenerator
n_{cell}	last cell

air rotary regenerator [22], later extended to a multi-objective optimization (pressure drop and effectiveness); optimum length of the heat exchanger and core porosity for maximum heat transfer rate per unit of frontal surface area [24]; the geometry of the matrix of a periodic-flow heat exchanger with the unit cost of the exergy of the warm delivered air as the objective function [25]; geometry and operational parameters for minimum entropy generation [26]; optimal dimensions for maximum energy performance [27]; optimum temperature efficiency of the regenerator providing minimum heat transfer surface area [28]; design features to enable best Stirling engine efficiency [29]; and the effect of design parameters on the exergetic efficiency of the regenerator [30,32]. All these works rely, of course, on some sort of numerical modeling of the regenerator. In this literature review, no previous studies on the optimization of regenerators heat exchangers applied to absorption refrigeration systems were found.

An experimental system was built in the laboratory using an absorption refrigerator and a car engine. The hot gas from the engine exhaust provides the waste heat used to drive the absorption refrigerator. Fig. 1 shows a photograph of the prototype and Fig. 2 shows the corresponding schematic diagram illustrating the main system components and energy interactions.

This paper introduces an unsteady mathematical model for a regenerative heat exchanger, which is part of heat driven refrigeration systems that use waste heat (e.g., hot exhaust gases). The main objective is to investigate the existence of optimum operating and geometric parameters for maximum system efficiency.

2. Mathematical model

The regenerator consists of a heat exchanger in which a metallic matrix is mounted inside the hot fluid passage to provide energy storage capabilities. The metallic matrix is particularly important when the flow rate of hot fluid is not enough to keep the system working at the desired levels of performance. In these situations, the amount of energy (heat) saved by the metallic matrix will

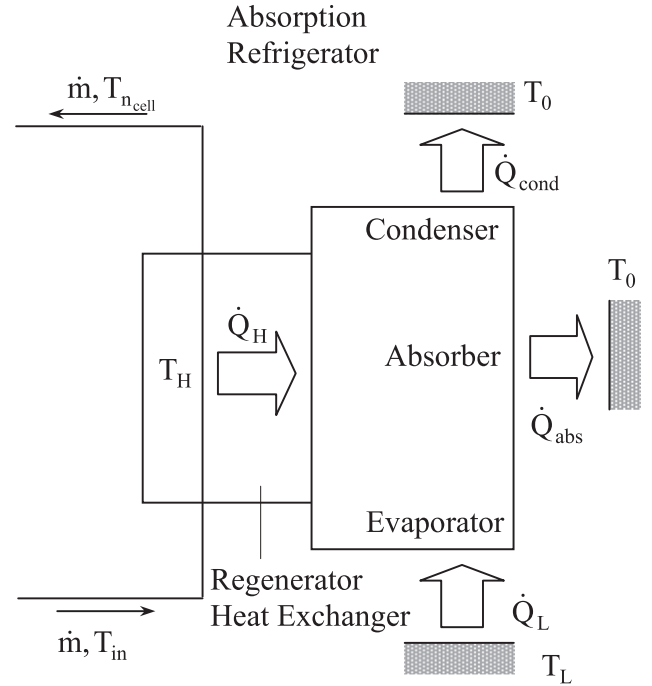


Fig. 2. Schematic diagram with the main components of the absorption refrigerator equipped with a regenerator heat exchanger.

sustain the amount of heat transferred to the cold fluid despite of the low hot fluid mass flow rate.

The amount of metallic matrix inside the hot fluid tube is represented in the model by the porosity, $\phi = V_V/V_T$.

2.1. Cold fluid without phase change

As a preliminary step, it is studied how the amount of metallic matrix affects the performance of the regenerator when there is no phase change of the cold fluid.

Fig. 3 shows the regenerator divided into small control volumes named “cells”, in which $L = n_{\text{cell}}\Delta x$.

Each cell of the regenerator is divided in three different systems. System 1: metallic matrix and tube; system 2: hot fluid, and system 3: cold fluid. Fig. 4 shows the thermal interactions among the systems in a generic cell “m” of the regenerator.

2.1.1. System 1: metallic matrix and tube

Applying the first law of thermodynamics to system 1, the resulting equations are

$$q_g + q_{\text{cond,p}} + q_{\text{cond,a}} + q_t = m_g^m c_g \frac{dT_g^m}{dt} \quad (1)$$

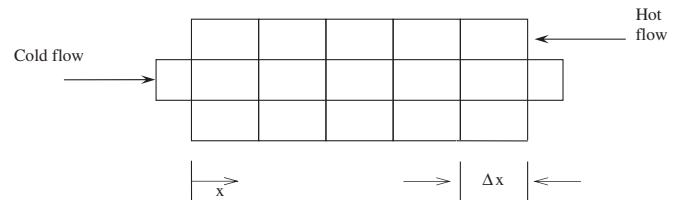


Fig. 3. Heat exchanger divided into control volumes.

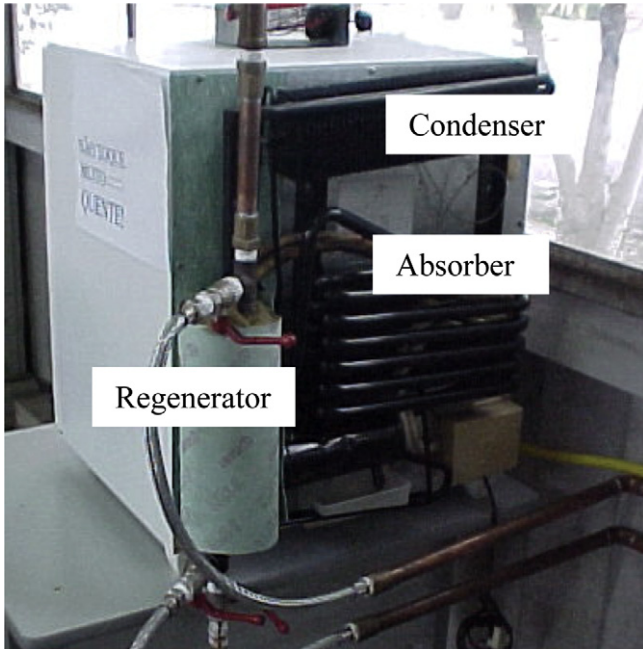


Fig. 1. Prototype of absorption refrigerator equipped with a regenerator heat exchanger.

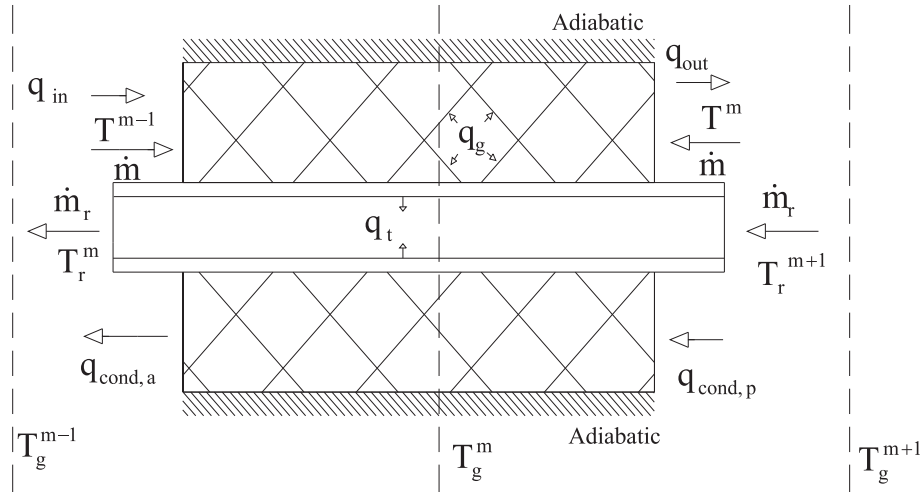


Fig. 4. Heat and mass transfer at cell "m".

where $q_g = h_g A_{gl} (T^m - T_g^m)$; $q_{cond,p} = -k A_{gs} ((T_g^m - T_g^{m+1}) / \Delta x)$, $q_{cond,a} = -k A_{gs} ((T_g^m - T_g^{m-1}) / \Delta x)$, and $q_t = h_t A_{tl} (T_r^m - T_g^m)$.

Porosity is an important parameter in the model because it affects the thermal response of the regenerator. It is possible to express the areas in Equation (1) as functions of porosity, as follows:

$$A_{tl} = \pi d_t L \quad (2)$$

$$A_{gl} = \frac{4(1 - \phi)V_T}{d_{ar}} \quad (3)$$

$$A_{gs} L = (1 - \phi)V_T \quad (4)$$

where $V_T = (\pi/4)(D^2 - d_t^2)L$. The regenerator is insulated, therefore when $m = 1$, $q_{cond,a} = 0$, and when $m = n_{cell}$, $q_{cond,p} = 0$.

In this study, the heat transfer coefficients by convection, h_g and h_t are taken as constants, but it is possible to account for their dependence on the Reynolds (Re) and Prandtl (Pr) numbers using empirical correlations [33].

The total amount of mass of metallic matrix at cell "m" is expressed as a function of total volume of metal in the regenerator, as follows:

$$m_g^m = \rho_g V_{metal} \quad (5)$$

where $V_{metal} = A_{gs} L$.

2.1.2. System 2: hot fluid flow

The energy balance for system 2 is

$$q_{in} - q_{out} - q_g = m^m c_v \frac{dT^m}{dt} \quad (6)$$

where $q_{in} = \dot{m} c_p T^{m-1}$ and $q_{out} = \dot{m} c_p T^m$.

The mass of hot fluid in the m th cell, m^m , is expressed as a function of porosity as follow

$$m^m = \rho \phi V_T \quad (7)$$

When $m = 1$ (first cell), $T^{m-1} = T_{in}$.

2.1.3. System 3: cold fluid flow

The energy balance for system 3 is

$$-q_t + \dot{m}_r c_{pr} (T_r^{m+1} - T_r^m) = m_r^m c_{vr} \frac{dT_r^m}{dt} \quad (8)$$

where $m_r^m = \rho_r (\pi d_t^2 / 4) L$.

Equations (1), (6) and (8) comprise a system of three differential equations to be integrated in time for each cell. Considering that n_{cell} is the number of cells, 3 n_{cell} is the total number of ordinary differential equations to be integrated in time for the entire regenerator.

Next, the possibility of cold fluid (refrigerant) phase change in the regenerator is included in the analysis.

2.2. Cold fluid with phase change

Fig. 5 shows the thermal interactions among the systems that occur in any cell "m" of the regenerator when the cold fluid experiences a phase change. In the end of the regenerator, the refrigerant (ammonia) has to be totally vaporized in order to go forward in the cycle; if the vaporization of ammonia is not complete, the remainder of liquid ammonia shall return to the beginning of the regenerator and pass through it again. Because of these features, one more flow is inserted in the regenerator, as shown in Fig. 5.

Hence, the model includes hereafter four different systems, i.e.: system 1: metallic matrix and tube; system 2: hot fluid; system 3: cold fluid in the inner tube (non-vaporized ammonia) and system 4: cold fluid in the innermost tube (liquid and vapor ammonia mixture).

Ammonia and impurities (water) flow through the innermost tube. Phase change occurs as a result of heat received from the hot fluid flow. Depending on geometric and operating conditions some amount of liquid ammonia might not vaporize. The liquid ammonia that did not vaporize, returns through the inner tube (system 3) to a reservoir located before the regenerator in order to re-enter the regenerator.

In the model, the water present in ammonia is assumed to be just a small fraction of the mixture (e.g., ejector refrigeration systems [8]), so that the refrigerant (ammonia) is treated a pure substance. If considered as a binary mixture, the temperature during phase change would no longer be constant, and the temperature variation should follow specific binary mixtures phase diagrams [34,35].

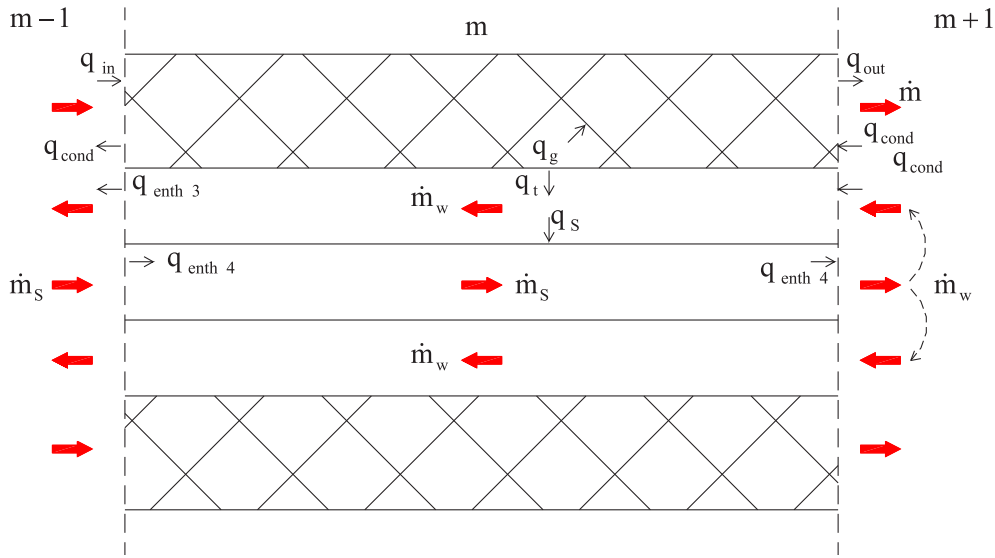


Fig. 5. Heat and mass transfer at cell “m” when there is phase changing of cold fluid (ammonia).

2.2.1. System 1: metallic matrix and tube

The equations developed for system 1 with no phase change of cold fluid can be used to model system 1 with phase change. This is possible because the mass and heat transfer that occur in system 1 do not depend on the ammonia phase change.

2.2.2. System 2: hot fluid flow

The energy balance for system 2 is

$$\dot{m}c_p(T^{m-1} - T^m) - q_g = \dot{m}c_v \frac{dT^m}{dt} \quad (9)$$

when $m = 1$ (first cell), $T^{m-1} = T_{in}$.

2.2.3. System 3: cold fluid flow in the inner tube (non-vaporized ammonia)

The energy balance for system 3 is

$$-q_t + q_s + q_{enth3} = \left(\dot{m}_{NH_3}^m c_{NH_3,l} + \dot{m}_{H_2O}^m c_{H_2O} \right) \frac{dT_w^m}{dt} \quad (10)$$

where

$$q_s = U_{34} A_{tl4}^m (T_s^m - T_w^m); q_{enth3} = \left(\dot{m}_{NH_3}^m c_{NH_3,l} + \dot{m}_{H_2O}^m c_{H_2O} \right) \cdot (T_{w,in}^m - T_w^m)$$

and

$$A_{tl4}^m = \frac{\pi \left(d_{ts} + \frac{e_4}{2} \right) L}{n_{cel}}.$$

U_{34} can assume two different values depending on whether there is phase change or not. When there is no phase change:

$$U_{34} = U_s = \frac{1}{\frac{1}{h_3} + \frac{e_4}{k_s} + \frac{1}{h_4}}$$

While ammonia is vaporizing the global heat transfer coefficient assumes numerical values greater than U_s . In order to express U_{34}

during phase change, a ratio between the global heat transfer coefficient when there are two phases in the system (vapor and liquid) to when just one phase is present (vapor or liquid), U_b was defined as follows:

$$U_b = \frac{U_{34}}{U_s} \quad (11)$$

The mass of liquid ammonia (if present) in system 3 at cell “m” can be expressed as a function of the quality of ammonia as follows:

$$m_{NH_3}^m = (1 - x_F^{n_{cell}})(1 - r)m_T^m \quad (12)$$

where r is the mass fraction of water in the strong solution defined as $r = \dot{m}_{H_2O}/\dot{m}_T$ and $\dot{m}_T = \dot{m}_{NH_3,s} + \dot{m}_{H_2O}$.

The total mass of refrigerant in cell “m” in the system 4 is

$$m_T^m = \frac{\Delta x A_s}{v} \quad (13)$$

where $v = (1 - r)v_{NH_3} + rv_{H_2O}$.

The mass of water in ammonia at cell “m” is

$$m_{H_2O}^m = r m_T^m \quad (14)$$

2.2.4. System 4: cold fluid flow in the innermost tube (liquid ammonia)

The energy balance for system 4 is

$$q_{enth4} - q_s = \left(\dot{m}_{NH_3,s}^m c_{NH_3,i} + \dot{m}_{H_2O}^m c_{H_2O} \right) \frac{dT_s^m}{dt} \quad i = (l, v) \quad (15)$$

While liquid ammonia is vaporizing the temperature is kept constant and equal to the saturation temperature. Hence, in this case:

$$\frac{dT_s^m}{dt} = 0 \quad (16)$$

Equation (16) is applied from the first cell “m” where the vaporization begins until the phase change is complete (or the last cell of the regenerator is reached).

In order to choose between Equations (15) and (16) to model the behavior of system 4, it is necessary to find out where the phase change starts and finishes or where there is no phase change. This can be done checking if $T_s^m \geq T_{sat}$ and if the mass of vapor of ammonia generated according to the flow is less or equal than the total mass of ammonia at cell “m”, i.e., $m_{NH_3,v}^m \leq (1-r)m_T^m$. If these two conditions are true, then the refrigerant at cell “m” is changing phase, otherwise the refrigerant is liquid (in the beginning of the regenerator) or vapor in the end of the regenerator.

The temperature T_s^m is calculated solving the Equation (15) or assuming that $T_s^m = T_{sat}$ during phase change.

As long as $T_s^m > T_{sat}$, $m_{NH_3,v}^m$ will remain constant and equals to zero. The mass of refrigerant that evaporates in each cell “m” is

$$\Delta m_{NH_3,v}^m = \frac{-q_s \Delta t}{h_{fg}(T_{sat})} \quad (17)$$

where $\Delta t = \Delta x/u$ and $u = \dot{m}_T v / A_s$.

The mass of refrigerant vapor accumulated in cell “m” is

$$m_{NH_3,v}^m = m_{NH_3,v}^{m-1} + \Delta m_{NH_3,v}^m \quad (18)$$

The next step is to calculate the quality of the regenerator in cell “m”, x_r^m .

$$x_r^m = \frac{m_{NH_3,v}^m}{(1-r)m_T^m} \quad (19)$$

When $m_{NH_3,v}^m > (1-r)m_T^m$ (or $x_r^m > 1$), the phase change is complete and Equation (16) does not express the behavior of system 4 anymore, then Equation (15) shall be used.

The mass of liquid ammonia or ammonia totally vaporized in cell “m” and the mass flow rate of ammonia in cell “m” are respectively $m_{NH_3,s}^m = (1-r)m_T^m$ and $\dot{m}_{NH_3,s} = (1-r)\dot{m}_T$.

The energy rate through system 4 in the cell “m” is

$$q_{enth4} = (\dot{m}_{NH_3,s} c_{NH_3,i} + \dot{m}_{H_2O} c_{H_2O}) \cdot (T_{s,in}^m - T_s^m) \quad (i = 1, v) \quad (20)$$

It is important to emphasize that in this model, the refrigerant is considered a pure substance because the mass fraction of water in the refrigerant is very small ($r = 0.01$). This value of mass fraction allows for the assumption that the temperature during phase change remains constant and equal to the ammonia saturation temperature.

The energy balance for system 4 is expressed by Equation (15) when there is one phase in the system (liquid or vapor) and it is expressed by Equation (16) when there is phase change.

2.3. Efficiency

The efficiency (η) of the regenerator is defined as the ratio between the actual heat transfer rate to ammonia vapor and the maximum possible heat transfer rate.

$$\eta = \frac{q_{NH_3,v}}{q_{max}} \quad (21)$$

where

$$q_{max} = \dot{m}_c p (T_{in} - T_{in,4}) \quad (22)$$

$$q_{NH_3,v} = (1-r)x_r^{n_{cell}} \dot{m}_T \left\{ c_{NH_3,i} (T_{sat} - T_{in,4}) + h_{fg}(T_{sat}) + c_{NH_3,v} (T_s^{n_{cell}} - T_{sat}) \right\} \quad (23)$$

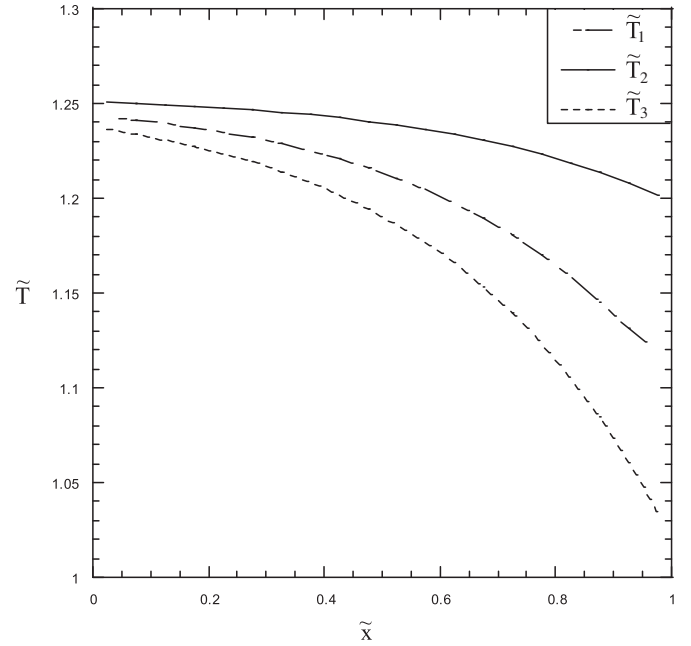


Fig. 6. Temperature distribution in the regenerator for $\phi = 0.95$.

The regenerator heat exchanger is modeled according to two different conditions. In the case where there is no phase change, the regenerator is represented in each cell by a system of three ordinary differential equations, Equations (1), (6) and (8); where T_0^m , $T_{g,0}^m$ and $T_{r,0}^m$ are the initial conditions. In the case where the phase change is present, the regenerator is represented in each cell by a system of four ordinary differential equations, Equations (1), (9), (10), (15) or (16); where T_0^m , $T_{g,0}^m$, $T_{w,0}^m$ and $T_{s,0}^m$ are the initial conditions. For both cases, the systems were solved using a Runge–Kutta 4th/5th order method [36].

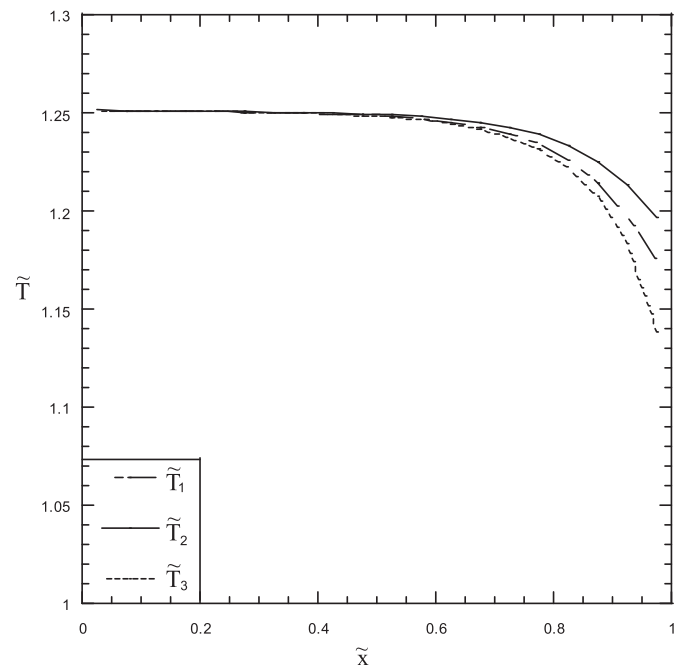


Fig. 7. Temperature distribution in the regenerator for $\phi = 0.5$.

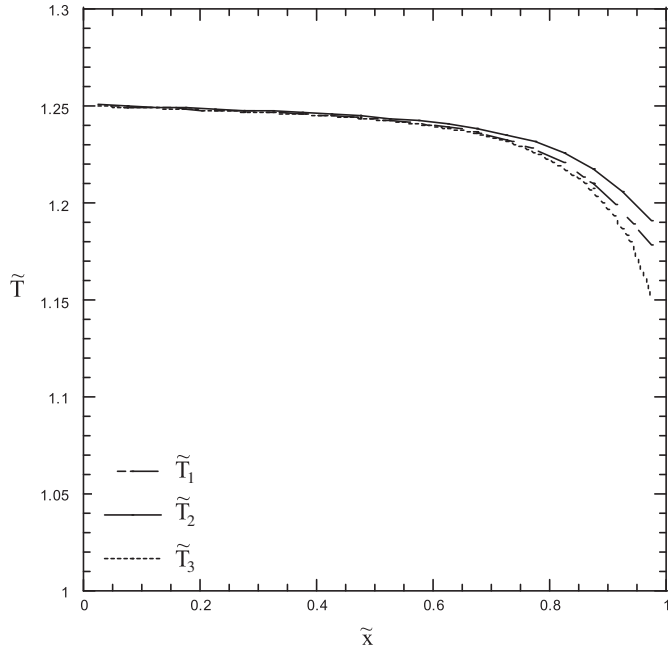


Fig. 8. Temperature distribution in the regenerator for $\phi = 0.2$.

The temperature of each system is obtained until the steady state is reached and after that the efficiency is calculated. The steady state is reached when the norm of the solution vector becomes independent of time according to a pre-specified tolerance ($\varepsilon = 10^{-4}$), as follows:

$$\frac{\|\vec{T}(t + \Delta t) - \vec{T}(t)\|}{\|\vec{T}(t)\|} < \varepsilon \quad (24)$$

and the number of cells in the mesh is taken as $n_{\text{cell},j}$ when the norm of the steady state solution vector does not vary between a less refined (j) and a more refined mesh ($j + 1$)

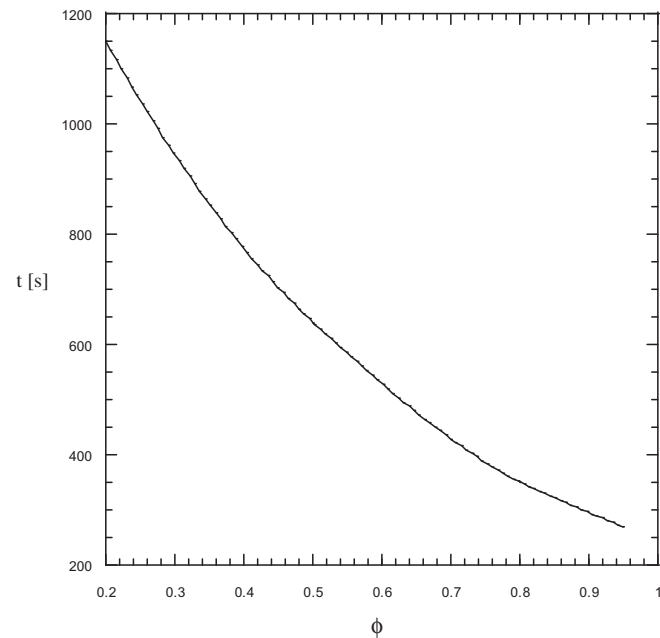


Fig. 9. Time to reach steady operation according to different porosities (ϕ).

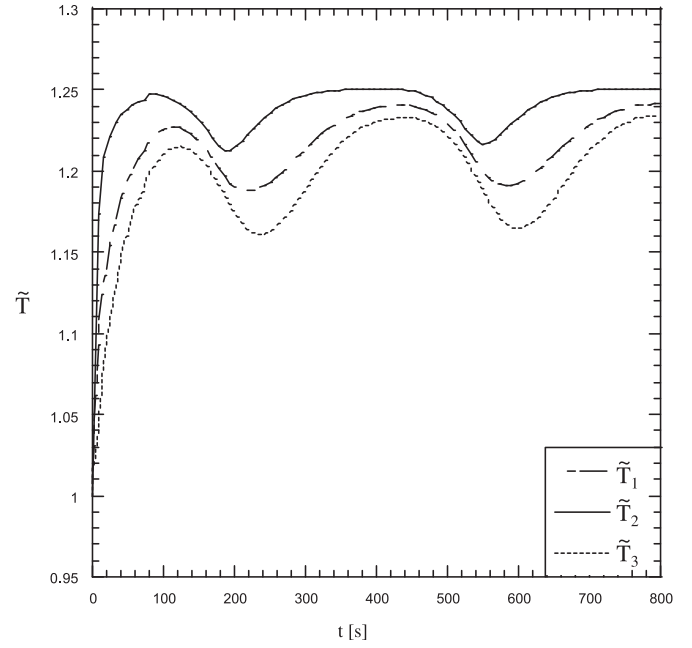


Fig. 10. Temperature as a function of time in the first cell when the hot fluid mass flow rate is modeled by a periodic function ($\phi = 0.95$).

$$\frac{\|\vec{T}_{j+1} - \vec{T}_j\|}{\|\vec{T}_j\|} < \varepsilon \quad (25)$$

3. Results and discussion

3.1. Model without cold fluid phase change

The initial conditions for this configuration are $T_0 = 373.15$ K, $T_{r,0} = 293.15$ K, $\dot{m} = 0.1$ kg s $^{-1}$ and $L = 1$ m. The numerical

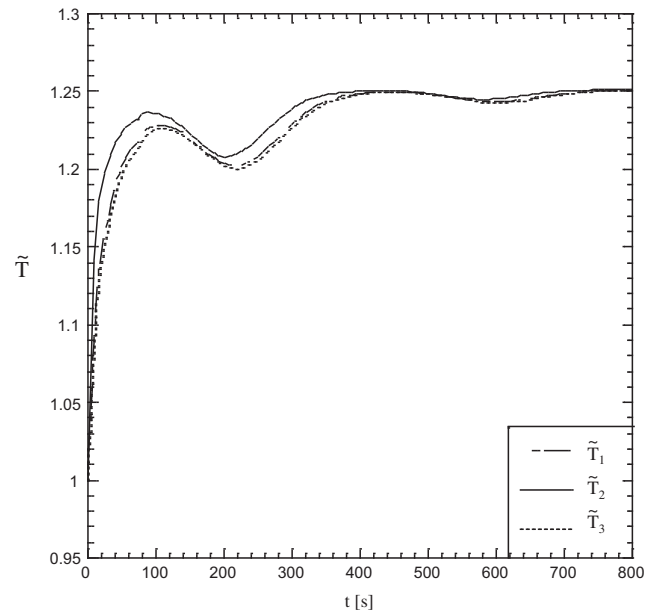


Fig. 11. Temperature as a function of time in the first cell when the hot fluid mass flow rate is modeled by a periodic function ($\phi = 0.5$).

Table 1

Physical properties and geometric data used in the numerical solution of the systems.

$c_g = 896 \text{ J kg}^{-1} \text{ K}^{-1}$
$c_p = 1000 \text{ J kg}^{-1} \text{ K}^{-1}$
$c_{pr} = c_{vr} = 4180 \text{ J kg}^{-1} \text{ K}^{-1}$
$c_v = 713 \text{ J kg}^{-1} \text{ K}^{-1}$
$d_{ar} = 0.002 \text{ m}$
$e_4 = 0.001 \text{ m}$
$h_3 = h_4 = 100 \text{ W m}^{-2} \text{ K}^{-1}$
$k = 14.9 \text{ W m}^{-1} \text{ K}^{-1}$
$\rho = 1165 \text{ kg m}^{-3}$
$\rho_g = 2707 \text{ kg m}^{-3}$

simulation of the regenerator is performed by solving Equations (1), (6) and (8). The scales $T_\infty = 298.15 \text{ K}$, $L_{\text{ref}} = 1 \text{ m}$, and $\dot{m}_{\text{ref}} = 1 \text{ kg s}^{-1}$ are used to nondimensionalize the temperatures, length of the regenerator and mass flow rates, respectively, so that $\tilde{T}_i = T_i/T_\infty$ ($i = 1, 2, 3, 4$), $\tilde{x} = x/L_{\text{ref}}$ and $\tilde{m} = \dot{m}/\dot{m}_{\text{ref}}$.

Figs. 6–8 show the numerically obtained dimensionless temperatures distribution in the regenerator heat exchanger for $\phi = 0.95, 0.5$, and 0.2 , respectively. It is seen that for $\phi = 0.2$ the curves are closer to each other than for $\phi = 0.95$ and 0.5 . This suggests that the heat transfer is more efficient when the porosity

is small (more metallic mass), since the more metallic mass the more heat transfer area results between system 1 and system 3, as it is physically expected.

The time to reach steady state operation as a function of porosity is shown in Fig. 9. The time to reach steady operation increases as the amount of metallic mass increases (porosity decreases) due to increasing thermal inertia, showing a trade off that has to be balanced in actual regenerator heat exchanger design.

When the curves in Fig. 8 are too close to each other to a point in which they can be represented by a single line, it is reasonable to assume that there is no significant amount of heat being transferred in that specific region of the heat exchanger, therefore it could be shortened. So, for the specific configuration shown in Fig. 8, the heat exchanger could be 0.4 m long instead of 1 m long. This feature is very important when space is constrained, as for example, in automobile air conditioning systems.

The next step is to study how the porosity affects the efficiency when the mass flow rate of hot fluid varies (due to some disturbance). In order to simulate this behavior a periodic function was selected to define the mass flow rate.

$$\dot{m} = \frac{\dot{m}_{\text{max}}}{2} \cos(t) + \frac{\dot{m}_{\text{max}}}{2} \quad (26)$$

For this analysis, the simulation was conducted until $t = 800 \text{ s}$. The results are shown in Figs. 10 and 11, for $\phi = 0.95$ and 0.5 , respectively. The goal is to determine the behavior of the regenerator when a disturbance occurs in the hot fluid mass flow rate. As it was concluded previously in the text, when the porosity decreases the temperature of the systems get close to each other due to a better thermal contact, however there is an additional and positive effect. When the porosity is low, the effect of the disturbance is better “absorbed” by the metallic mass than with a higher porosity. Therefore the regenerator thermal response is more robust as porosity decreases, i.e., the system becomes less subject to perturbations such as fluctuations in heat supply, for example.

3.2. Model with cold fluid phase change

The initial conditions for this part of the analysis are the same as those used when there is no cold fluid phase change, but the

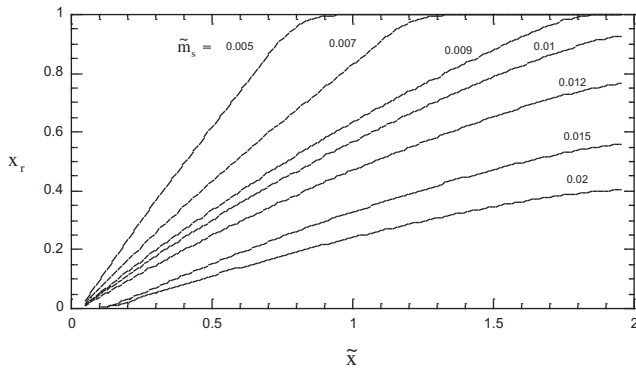


Fig. 12. Quality distribution in the regenerator for $\phi = 0.5$.

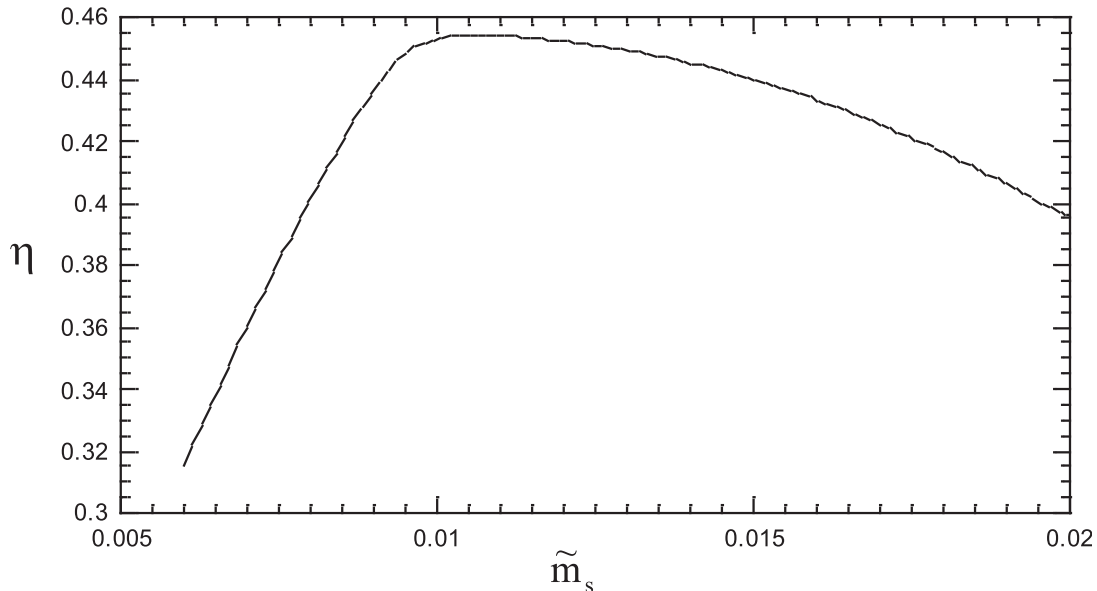


Fig. 13. Efficiency as a function of dimensionless hot fluid mass flow rate for $\phi = 0.5$.

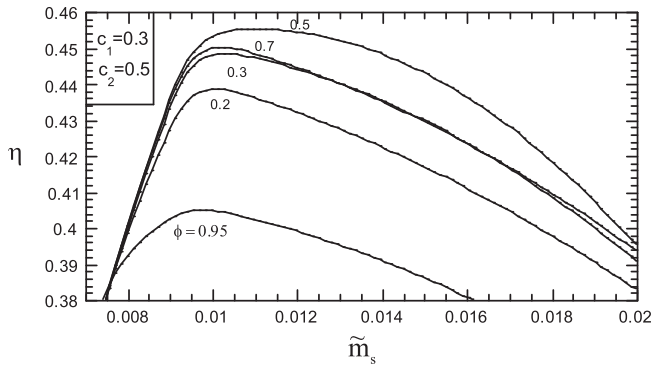


Fig. 14. Efficiency as a function of dimensionless hot fluid mass flow rate for different porosities.

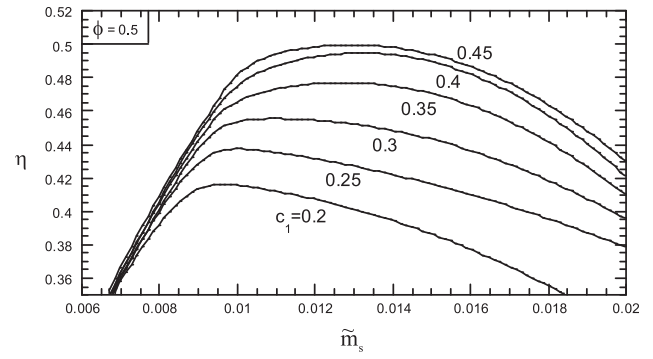


Fig. 16. Efficiency of the regenerator for different values of c_1 ($c_2 = 0.5$).

modeling equations are different. The numerical simulation of the regenerator is performed by solving Equations (1), (9), (10), and (15) or (16).

The geometric ratios $c_1 = d_{ts}/D$ and $c_2 = d_{in}/D$ are defined. The results for the physical properties and geometric data listed in Table 1 are plotted in Figs. 12 and 13, for $\phi = 0.5$, $c_1 = 0.3$ and $c_2 = 0.5$. In Fig. 12, the variation of ammonia quality, x_r , with respect to regenerator dimensionless length, $\tilde{x}$, is represented as a function of different liquid ammonia dimensionless mass flow rates, $\tilde{m}_s$. It is noted that, for $\tilde{m}_s > 0.009$ the ammonia flow does not vaporize completely, as it is physically expected since the heat input rate is the same as mass flow rate increases. Fig. 13 shows that the efficiency displays a maximum for $\tilde{m}_{s,opt} \sim 0.01$, i.e., the configuration with higher efficiency is the one that totally vaporizes the ammonia flow and uses the entire heat exchanger length for that, delivering ammonia as saturated vapor at the outlet, i.e., neither as superheated vapor nor as a two-phase mixture (liquid and vapor).

The main objective of the study is to find possible optimal configurations leading to the most efficient regenerator. Therefore, besides the dimensionless liquid ammonia mass flow rate, in Fig. 14, the effect of porosity on efficiency is investigated. The efficiency of the regenerator is depicted as a function of $\tilde{m}_s$ for different values of porosity, when $c_1 = 0.3$ and $c_2 = 0.5$. According to Fig. 14 the now two-way maximized efficiency is 45.5% for the optimal pair $(\phi, \tilde{m}_s)_{opt} = (0.5, 0.01)$.

The existence of the optima found with respect to $\tilde{m}_s$ in Fig. 14 is explained by the analysis of two extremes. For a fixed heat input rate, when the mass flow rate of ammonia is very large (approaches infinity), liquid ammonia mass flow rate to be heated is too high, thus the difference between inlet temperature and outlet temperature is small, hence the actual heat transfer rate to ammonia vapor, $q_{NH_3,v}$, is also small. When the

mass flow rate of ammonia approaches zero, $q_{NH_3,v}$ is also small, because it is proportional to $\tilde{m}_s$. Analogously, when $\phi \rightarrow 0$, the empty area available for the hot fluid to flow decreases so that the velocity of hot fluid increases. This effect leads to inlet and outlet temperatures with very similar values making the efficiency to decrease. When $\phi \rightarrow 1$, the heat transfer area between the hot fluid and metallic matrix decreases, so that the efficiency also decreases.

Fig. 15 summarizes the two-way optimization procedure showing that the one-way maximized efficiency, η_{max} , with respect to $\tilde{m}_s$, depicts a second maximum with respect to porosity, ϕ . Furthermore, $\tilde{m}_s \sim 0.01$ for all porosities, i.e., the optimal operating condition is “robust” with respect to porosity variation, which is an important feature for regenerator operation.

3.3. Parametric analysis

Fig. 16 shows the results obtained when the porosity is kept constant ($\phi = 0.5$), $c_2 = 0.5$ and the values of c_1 are varied. It is observed that there is no optimal c_1 , i.e., the efficiency increases as c_1 increases. This feature occurs because in the model, the pressure drop was neglected in the pipes of the regenerator. In the optimization of the geometric configuration of the regenerator the pressure drop is therefore an important feature to be considered.

4. Conclusion

In this paper, a mathematical model for a regenerative heat exchanger was developed based on the volume element methodology to capture expected physical trends. Next, transient simulations were conducted and a thermodynamically optimized regenerator heat exchanger design was proposed.

This study demonstrated that the insertion of a metallic matrix (represented by a porosity parameter in the model) helps the system to become more stable to possible disturbances in the mass flow rate supply of hot fluid.

The model was then used to determine optimal parameters for a specific geometric configuration of the regenerator, finding that the two-way maximum efficiency operating with optimal porosity and dimensionless mass flow rate of liquid ammonia, is $\eta_{max,max} = 45.5\%$ for the optimal pair $(\phi, \tilde{m}_s)_{opt} = (0.5, 0.01)$. The main conclusion is that it is expected that, after experimental validation and model adjustment to ensure the accuracy of results, the developed mathematical model could be used as an efficient tool for the thermal design, control and optimization of regenerator heat exchangers with similar characteristics to the heat exchanger studied in this paper.

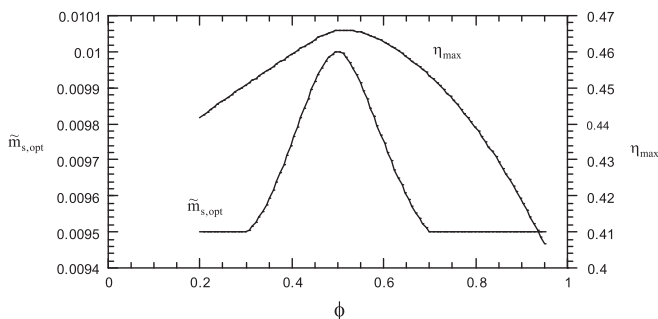


Fig. 15. Optimal hot fluid mass flow rate and efficiency as functions of porosity.

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